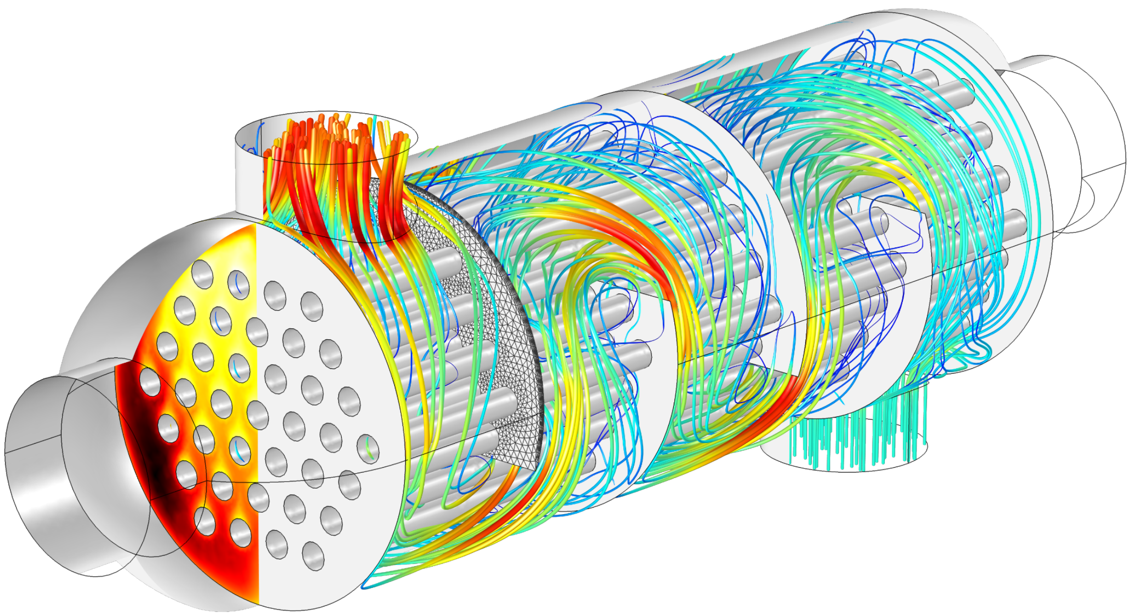
Shell and Tube Heat Exhanger



Group - 3
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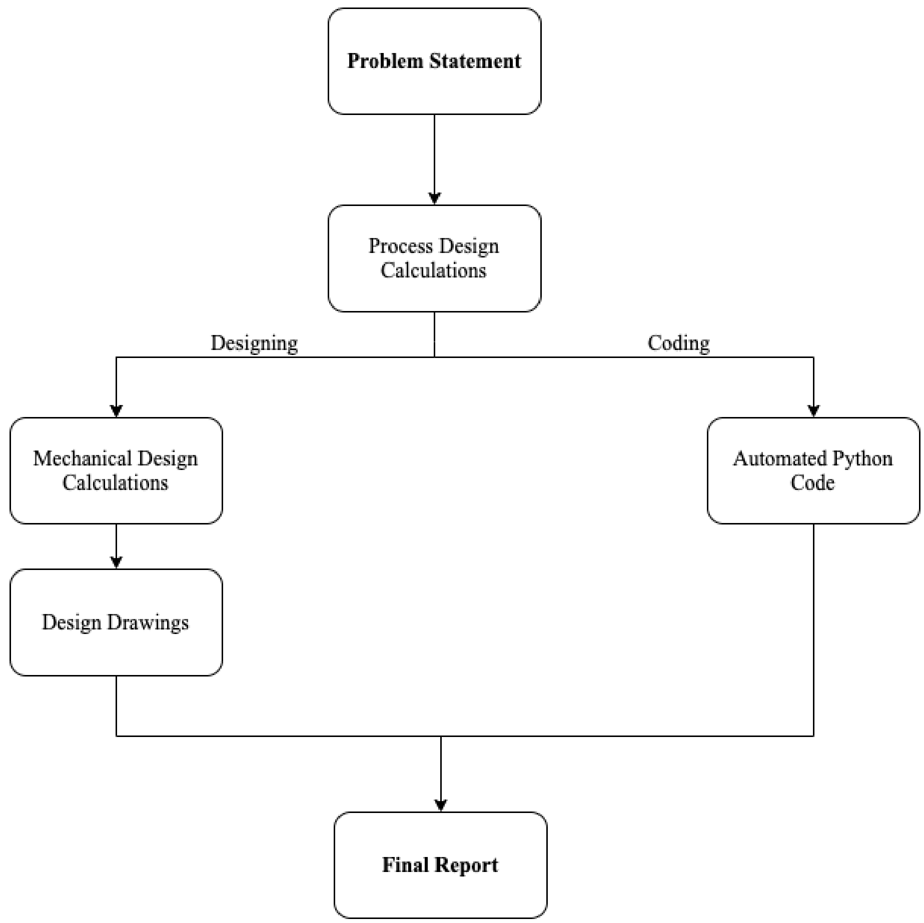
Flow Chart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Problem Statement

Design a Shell and Tube Heat Exchanger (stripped Heavy Naptha Trim cooler) to cool heavy naphtha of 1.5+0.03*(Group Number) kg/s at 85^0C to 55^0C using water at 35^0C, as the coolant

Shell and Tube Heat Exchanger

Fluid Allocation

Heavy naptha is considered on the tube side, since the viscosity of heavy naptha is more than that of water. Heavy naptha is more corrosive than water, hence choosen as the tube side liquid, as shell is more expensive.

Tube Side fluid : Heavy Naptha
Shell Side fluid : Water

Process requirements

1. Design Tolerance < 5%
2. Tube side Pressure Drop < 10 psi
3. Overdesign Area < 10%

Given, mass flow rate of heavy naptha $= 1.5 + (0.03)(29)$
$= 2.37 \ kg/s$

For water, $T_{c,in} = 35°c$ For Heavy Naptha, $T_{h,in} = 85°c$

$T_{c,out} = 45°c$ $T_{h,out} = 55°c$

API of heavy naptha $= 47.3°$
API of water $= 10°$

Calarofic Temperatures

$$r = \frac{\Delta t_c}{\Delta T_H} = \frac{T_{h,out} - T_{c,in}}{T_{h,in} - T_{c,out}} = \frac{55-35}{85-45}$$

$$\therefore r = 0.75$$

From D.Q. Kern Pg : 827,

For water, at $r = 0.75$, $K_c = 1.24$ (By extrapolation)

For heavy naptha, at $r = 0.75$, $K_c = 0.1$

$\therefore$ F_c for heavy naptha $= 0.44$
F_c for water $= 0.38$

$$T_h^* = T_{h,out} + F_c (T_{h,in} - T_{h,out}) = 55 + 0.44(85-55)$$
$$= 67°c$$

$$T_c^* = T_{c,in} + F_c (T_{c,out} - T_{c,in}) = 35 + 0.38(45-35)$$
$$= 38.8°c$$

Properities of Fluids at Calarofic Temperatures;*

Heavy Naptha

Specific Heat Capacity @ $T = T_h^* (C_{p_h}) = 2036.9947$ J/kg K
Viscosity @ $T = T_h^* (\mu_h) = 0.733$ cP
Density @ $T = T_h^* (\rho_h) = 791.387$ kg/m³
Thermal Conductivity @ $T = T_h^* (k_h) = 0.15$ W/m K

Water

Specific Heat Capacity @ $T = T_c^* (C_{p_c}) = 4178.5761$ J/kg K
Viscosity @ $T = T_c^* (\mu_c) = 0.6681$ cP
Density @ $T = T_c^* (\rho_c) = 992.6812$ kg/m³
Thermal Conductivity @ $T = T_h^* (k_c) = 0.598$ W/m K

* All the values have been computed using the python code using, by fitting a polynomial and finding the relation between property and temperature using data found online (Refer to "find_prop" fn for more, all the data references have been attached in the code)

Enthalpy Balance

$$\dot{m}_h C_{p_h} (T_{h,in} - T_{h,out}) = \dot{m}_c C_{p_c} (T_{c,out} - T_{c,in})$$

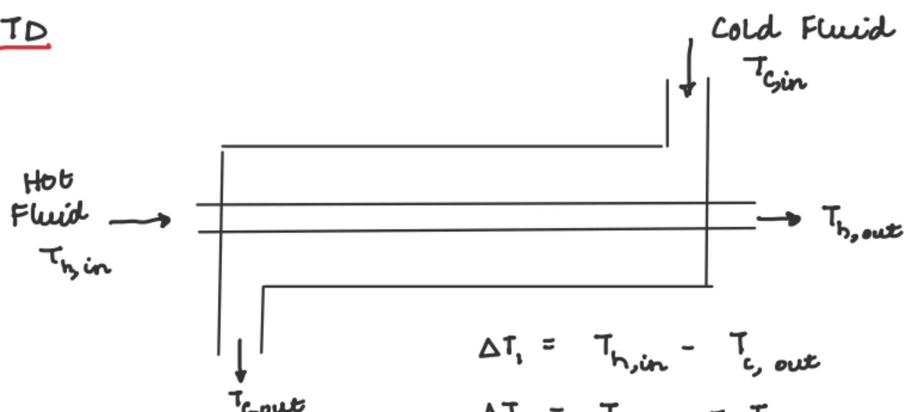
$$\Rightarrow 2.37 \times 2036.9947 \times 30 = \dot{m}_c \times 4178.5761 \times 10$$

$$m_c = \frac{2.37 \times 2036.9947 \times 30}{4178.5761 \times 10}$$

$$\therefore m_c = 3.466 \text{ kg/s}$$

Heat duty : $Q = 2.37 \times 2036.9947 \times 30$
$= 144830.3232$ J/s

LMTD



$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

$$\therefore \text{ LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{40-20}{\ln(2)} = 28.8539^\circ C$$

LMTD Correction Factor

$$R = \frac{(T_{h,in} - T_{h,out})}{(T_{c,out} - T_{c,in})} = \frac{20}{10} = 2$$

$$S = \frac{(T_{c,out} - T_{c,in})}{(T_{h,in} - T_{h,out})} = \frac{10}{30} = \frac{1}{3}$$

For 1-2 heat exchanger : $F_T = 0.935$ (D.Q. Kern, Pg: 828)

We proceed with this value as $F_T > 0.9$

Assumed U

Overall $U_D \in (50, 125)$ BtU/hr ft² °F
(Since heavy naptha is a medium organic)

Let $U_{assm} = \left(\frac{50+125}{2}\right) = 87.5$ BtU/hr ft² °F
$= 496.848$ BtU / hr ft² °F

$\dot{Q} = U A F_T \text{ (LMTD)}$

$$A = \frac{\dot{Q}}{U F_T \text{ LMTD}} = \frac{144830.3232}{496.848 \times 0.935 \times 28.8539}$$

$\therefore$ $A = 10.8043 \text{ m}^2$

Number of Tubes

$$A = (\pi d_o L)\, n_t$$

Consider, Tube length (L) = 20 ft = 6.096 m

Consider the standard tube diameters,

d_i = Tube ID = 0.67 inch = 0.017018 m (BWG=8)
d_o = Tube OD = 1 inch = 0.0254 m

$$\therefore n_t = \frac{10.8043}{\pi \times 0.0254 \times 6.096} \simeq 23 \text{ tubes}$$

Standard Number of Tubes available = 32 (D.Q Kern, Pg: 841)

Number of Tube passes (N_t) = 2
Number of shell passes = 1

We select a square pitch over triangular pitch, as,
(i) Cleaning is easier
(ii) Lower Pressure drop
(iii) n_a isn't too high

Shell ID = 10 inch = 0.0254 m
Square Pitch = $1\frac{1}{4}$ inch = 0.03175 m

Tube Side Reynold's Number

$$Re_{tube} = (N_t \times \dot{m}_h) / \left(\frac{\pi}{4} \times d_i \times \mu_h \times n_t\right)$$

$$= \frac{2 \times 4 \times 2.37 \times 1000}{0.017018 \times 0.733 \times 32} = 15112.8485$$

Tube side heat transfer co-efficient

Dittus Boiter Eqn:, $Nu = 0.023\ Re^{0.8}\ Pr^{1/3} \left(\frac{\mu_w @ T = T_c^*}{\mu_w @ T = T_h^*}\right)$

$$\frac{h_i d_i}{k_h} = 0.023 \left(\frac{C_p \mu}{k_h}\right)^{1/3} Re^{0.8} \left(\frac{\mu_w @ T_c^*}{\mu_w @ T_h^*}\right)$$

$$\Rightarrow h_i = 0.023 \times \left(\frac{2036.9947 \times 0.733}{0.1500 \times 1000}\right)^{1/3} (15112.8485)^{0.8} \times \left(\frac{\mu_w'}{\mu_w''}\right)$$

$$\therefore h_i = 890.915\ W/m^2K$$

Shell side heat transfer co-efficient,

Important considerations, a) 25% Cut Segmental Baffles
b) Baffle spacing, $B = D_s/2$

$D_s = 0.254$ m, $P_T = 0.03175$ m
Tube clearance, $(C) = P_T - d_o = 0.00635$ m

$$a_s = \frac{CBD_s}{P_T} = 6.415 \times 10^{-3}\ m^2$$

$$D_{eq} = \frac{4\left(P_T^2 - \frac{\pi}{4} d_o^2\right)}{\pi d_o} = 0.0251\ m$$

$$Re_{shell} = \frac{\dot{m}_c D_{eq}}{a_s \mu_c} \quad \left(\text{i.e } Re_s = \frac{\rho v_s d_e}{\mu},\ v_s = \frac{\dot{m}}{\rho}\left(\frac{1}{a_s}\right)\right)$$

$$= \frac{3.466 \times 0.0251 \times 1000}{6.415 \times 10^{-3} \times 0.6681} = 20298.50941$$

$$\therefore Re_{shell} = 20298.50941$$

$R_{do} \equiv$ Tube Outside Dirt Factor, $R_{do} = 0.00075\ m^2K/W$

$R_{di} \equiv$ Tube Inside Dirt Factor, $R_{di} = 0.00018\ m^2K/W$

Tube material : Carbon steel, $k_{wall} = 54$ W/mK

Colbern Factor, $j_H = 79.6518$ (At $Re = Re_{shell}$)

$$j_H = \frac{h_o D_e}{k_c} (P_r)^{-1/3} \left(\frac{\mu_\infty}{\mu_{wall}}\right)^{-0.14}$$

$$\Rightarrow h_o = 1134.0369 \text{ W/m}^2\text{K}$$

Using, $$\frac{1}{U_{cal}} = \left[\frac{1}{h_o} + R_{do} + \frac{A_o}{A_i}\left(\frac{d_o - d_i}{2 k_{wall}} + R_{di} + \frac{1}{h_i}\right)\right]$$

Substituting and evaluating, $U_{cal} = 244.7841$ W/m²K

$$\text{Relative Error} = \left|\frac{U_{cal} - U_{assm}}{U_{assm}}\right| \times 100$$
$$= \left|\frac{244.7841 - 496.848}{496.848}\right| \times 100$$
$$= 50.7326\%$$

But the tolerence is 5%, since our rel error > tolerence we perform further iterations

Iteration - 1

$U_{assm} = U_{cal} = 244.7841$ W/m²K

$A = \dot{Q}/(U_{assm} \times F_T \times LMTD) = 21.93$ m²

$n_t = A/\pi d_o L = 44$

Standard Number of Tubes available $(n_t) = 45$

$P_T = 1\frac{1}{4}$ inch, square layout

$D_s = 0.3048$ m

$Re_{tube} = 10746.9145$

$$h_i = \frac{k_n}{d_i} \times 0.023 \times (Re_{tube})^{0.8} (P_r)^{1/3} \left(\frac{\mu_\infty}{\mu_{wall}}\right)^{0.14}$$

$\therefore h_i = 678.24$ W/m²K

$B = D_s/2 = 0.1524$ m, $a_s = \dfrac{C B D_s}{P_T} = 0.00929$ m²

$Re_{shell} = 14034.8241$

Colbern Factor, $j_H = 65.54$ (at $Re = Re_{shell}$)

$$j_H = \frac{h_o D_e}{k_c} (P_r)^{-1/3} \left(\frac{\mu_\infty}{\mu_{wall}}\right)^{0.14} \Rightarrow h_o = 1185.542 \text{ W/m}^2\text{K}$$

$\therefore U_{cal} = 214.0128$ W/m²K

Relative error = 12.5708% (> 5%, hence we re-iterate)

Iteration - 2

$U_{assm} = U_{cal} = 214.0128 \; W/m^2K$

$A = \dot{Q}/(U_{assm} \times F_T \times LMTD) = 25.083 \; m^2$

$n_t = A/\pi d_o L = 52$

Standard Number of Tubes available (n_t) = 56

$D_s = 0.3365 \; m$

On solving, $Re_{tube} = 8635.91$

$$h_i = \frac{k_b}{d_i} \times 0.023 \times (Re_t)^{0.8} \times (Pr)^{1/3} \times \left(\frac{\mu_b}{\mu_{wall}}\right)^{0.14}$$

$\therefore h_i = 569.3847 \; W/m^2K$

$a_s = \frac{CBD_s}{P_T} = 0.0113 \; m^2 \Rightarrow Re_{shell} = 11511.6535$

Colburn Factor, $j_H = 58.95$ (at $Re = Re_{shell}$)

$\therefore h_o = 1066.4834 \; W/m^2K$

On solving, $U_{cal} = 189.83 \; W/m^2K$

Relative error = 11.3% (> 5%, hence we re-iterate)

Iteration - 3

$U_{assm} = U_{cal} = 189.83 \; W/m^2$

$A = \dot{Q}/(U_{assm} \times F_T \times LMTD) = 28.2787 \; m^2$

$n_t = A/\pi d_o L = 59$

Standard Number of Tubes availabe (n_t) = 76

$D_s = 0.3873 \; m$

On solving, $Re_{tube} = 6363.3046$

$h_i = 445.9697 \; W/m^2K$

$a_s = \frac{CBD_s}{P_T} = 0.15 \; m^2 \Rightarrow Re_{shell} = 8690.2$

Colburn Factor, $j_H = 50.74$ (At $Re = Re_{shell}$)

$\therefore h_o = 918.0437 \; W/m^2K$

On solving, $U_{cal} = 159.1453 \; W/m^2K$

Relative error : 16.1636% (>5%, hence we re-iterate)

Iteration - 4

$U_{assm} = U_{cal} = 159.83\ W/m^2$

$A = \dot{Q}/(U_{asm} \times F_T \times LMTD) = 33.7308\ m^2$

$n_t = A/\pi d_o L = 70$

Standard Number of Tubes available $(n_t) = 76$

$D_s = 0.3873\ m$

On solving, $Re_{tube} = 6363.3046$, $h_t = 445.9697\ W/m^2K$

$Re_{shell} = 8690.2$, $h_i = 918.0437\ W/m^2K$

$\therefore U_{cal} = 159.1453\ W/m^2K$

Relative error = 0%

$\therefore$ The overall heat transfer coefficient of 1-2 shell and tube heat exchanger is $U_d = 159.1453\ W/m^2K$

Pressure Drop

$n_t|_{final} = 76$

$Re_{tube} = 6363.3046$

From moody diagram, $f = 0.045$

Tube side velocity, $v_t = \dfrac{N_t\ \dot{m}}{n_t\ (\rho A)} = 0.3465\ m/s$

Tube side pressure drop,

$$\Delta P_f = \frac{1}{2} f \rho u^2 \left(\frac{L}{d_i}\right) = \frac{1}{2} \times 0.045 \times 791.39 \times 0.35^2 \times \left(\frac{20}{1}\right)$$
$$= 759.182\ Pa$$

$$\Delta P_r = \frac{4N_t}{S}\frac{v^2}{2g} = \frac{4\times 2}{791.39} \times \frac{0.35^2}{2\times 9.81} \times 1000$$
$$= 0.06187\ Pa$$

$$\Delta P_{total} = \Delta P_f + \Delta P_r$$
$$= 759.2446\ Pa$$

$\therefore \Delta P_{total} = 0.1101\ psi\ (< 10\ psi)$

Overdesign

$$\% \text{ Overdesign} = \left| \frac{A - A_{eq}}{A_{eq}} \right| \times 100$$

$$= \left| \frac{70-76}{76} \right| \times 100 = 8.5714\%$$

$\%$ Overdesign = 8.5714% ($< 10\%$)

Hence, the above design is acceptable and cost-effective.

Mechanical Design

Shell side passes = 1, Tube side passes = 2
No. of tubes = 76
Tube dimensions, $L = 20$ ft ($= 6.096$ m)
$d_i = 17.02$ mm ($= 0.017$ m)
$d_o = 25.4$ mm ($= 0.0254$ m)

Tube Material : Carbon Steel (IS: 4503 - 1967, Pg 22)
Pitch : Square Pitch

Shell diameter, ID = 387.3 mm
Shell and Head : Carbon Steel Material,
Torispherical Head

Carbon Steel Corrosion Allowance in Petroleum / chemical industeries where severe conds. are expected (C) = 3 mm

Design Temperature = $1.1 \times T_{max}$

$T_{max} = T_{h,i} = 85°C = 185°F$
$T_{design} = 1.1 \times T_{max} = 1.1 \times 185°F = 203.5°F$

$\therefore T_{design} = 203.5°F$

Design Pressure = $1.1 \times P_{operating}$

$P_{op} = 50$ psia $\Rightarrow P_{design} = 1.1 \times 50 = 55$ psia
$= 379.212$ kPa $= 0.38$ N/mm^2

Permissable Stress, $f = 100.6$ N/mm^2 (Carbon steel)

Shell Thickness Calculations

$P = 0.38\ N/mm^2$
$D_s = 387.3\ mm$
$f = 100.6\ N/mm^2$
$J = 0.8$ (Joint efficiency)

$$t_s = \frac{PDs}{fJ - 0.6P} + C = \frac{0.38 \times 387.3}{100.6 \times 0.8 - 0.6 \times 0.38} + 3 = 4.833\ mm$$

$\therefore\ t_s = 5mm$ [IS : 4503 - 1967, Pg : 22, Chemical / Petroleum Ind]

Shell Cover

Crown Radius, $R_i = D_s = 387.3\ mm$
Knuckle Radius, $r_i = 6\%\ R_i = 23.238\ mm$

Inside depth of head (h_i) : $R_i - \sqrt{\left[\left(R_i - \frac{D_s}{2}\right)\left(R_i + \frac{D_s}{2}\right) + 2r_i\right]}$

$$h_i = 387.3 - \sqrt{\left[(387.3)^2 - (387.3/2)^2 + 2(23.238)\right]}$$
$$= 51.8191\ mm$$

Effective exchanger length = $L_t + 2h_i$
$= 6.096 + 2\ (51.8191/1000)$
$= 6.1996\ m$

$\therefore\ L_{eff} = 6.1996\ m$

$$W = \frac{1}{4}\left(3 + \sqrt{\frac{R_i}{r_i}}\right) = 1.771$$

For head design, $J = 1$

$$t_h = \frac{PR_iW}{2fJ - 0.2P} + C = \frac{387.3 \times 0.38}{2 \times 1 \times 100.6 - 0.2 \times 0.38} + 3$$
$$= 3.731\ mm$$

Standard available thickness, $t_h = 5mm$

D_c = Outside Shell dia = $D_s + 2t_h$ = 397.3 mm

$C_1 = 0.3$ (Cover is bolted with narrow faced or ring type gaskets)

$P = 0.38\ N/mm^2 = \frac{0.38}{9.8} = 3.87492\ kgf/cm^2$

$f = 100.6\ N/mm^2 = 10.258\ kgf/mm^2$

$$t_{cc} = \frac{D_c}{10}\sqrt{\frac{C_1 P}{f}} + C = \frac{397.3}{10}\sqrt{\frac{0.3 \times 3.87}{10.258}} + 3$$
$$= 7.1782\ mm$$

Standard available thickness, t_{cc} = 10mm

Thickness of Pass Partition Plate

Since, $D_c < 600$ mm

Thickness of the pass partition plate = 10 mm (including corrosion allowance)

$\therefore\ t_{ppp} = 10$ mm

Tube Sheet Thickness

$F = 1$ (Fixed tube sheet) → supported tube sheets TEMA Pg: 55

$G_p = D_s = 387.3$ mm

$f = 100.6\ N/mm^2$

$$K = 1 - \frac{0.785}{\left(\frac{P_T}{d_o}\right)^2} = 0.4976$$

P = Effective Pressure $= (P_s + \cancel{P_b}^{\,0})$ or $(P_t + \cancel{P_b}^{\,0})$
$= P_s$ or $P_t = 0.38\ N/mm^2$

$\frac{P}{f} = 3.77 \times 10^{-3}$ — ①

$1.6\left(1 - \frac{d_o}{P_T}\right)^2 = 64 \times 10^{-3}$ — ②

$$\Rightarrow\ 1.6\left(1 - \frac{d_o}{P_T}\right)^2 > \frac{P}{f}$$

Shear stress won't control the tube sheet thickness.
→ Only 'Resist Bending' will be considered.

Min. Tube-sheet thickness for resist bending.

$$t_{ts} = \frac{F G_p}{3}\sqrt{\frac{P}{kf}} = \frac{387.3}{3}\sqrt{\frac{0.38}{0.4976 \times 100.6}}$$

$$t_{ts} = 11.248 \text{ mm}$$

Shell dia $\leq$ 500 mm

$$T_{design} = \max(t_{ts}, D_s/10)$$

$$\therefore T_{design} = \max(11.248, 38.73) = 38.73 \text{ mm}$$

Corrosion Allowance = 38.73 + 3 = 41.73 mm

Standard available Thickness = 45 mm

Impingment Plate

$$\rho = \rho_{h}|_{T_c^*} = 791.387 \text{ kg/m}^3 = 0.791 \text{ g/cm}^3$$

Nozzle Diameter = D_n = 76.2 mm = 3 inch (Nptel Table 2.3 Pg : 11)

Linear Velocity of Tube Fluid

$$u_{tube} = \frac{\dot{m}_h}{\left(\frac{\pi D_n^2}{4}\right)\rho_h} = \frac{2.37}{\left(\frac{\pi \times 0.076^2}{4}\right)791.387} = 0.6567 \text{ m/s}$$

$$\rho u^2 = 0.3413 \text{ g/cm}^3\text{s} \lll 125 \text{ g/cm}^3\text{s}$$

∴ Protection against impingment isn't required

Nozzle Thickness (t_n):

D_n = 76.2 mm
J = 0.8
f = 100.6 N/mm², P = 0.38 N/mm²

$$t_n = \frac{P D_n}{2fJ - P} + c = 0.18032 + 3$$

$\therefore\ t_n = 3.18032$ mm , Standard available thickness = 4mm

$P_s = P_t \Rightarrow (t_n)_{shell} = (t_n)_{tube}$

Design of Gaskets

Flat iron Jacketed, asbestos fill

$\Rightarrow$ Gasket factor (m) = 3.75

Maximum design seating stress (Y) = 5.35 kgf/mm²

$$Y = 52.4656\ N/mm^2$$

$$\frac{D_{OG}}{D_{IG}} = \sqrt{\frac{Y - Pm}{Y - P(m+1)}} = 1.05223$$

Outside Gasket Diameter (D_{OG})

Inside Gasket Diameter (D_{IG}) = $D_S + 0.25$
= 387.55 mm

$$D_{OG} = 1.05223 \times D_{IG} = 407.792\ mm$$

Gasket Width (N) = $\frac{D_{OG} - D_{IG}}{2}$ = 10.1209 mm
(Using 35 mm)

Mean Gasket Diameter (G) = $\frac{D_{OG} + D_{IG}}{2}$ = 397.671 mm

Basic Gasket Seating width (b_0) = $\frac{N}{2}$ = 17.5 mm

Eff. Gasket seating width (b) = $\frac{\sqrt{b_0}}{2}$ = 2.092 mm
($b_0 > 6$mm)

Bolts

G = 397.671 mm

Bolt load due to gasket rn. under atm. conditions,

$$W_{m_1} = \pi b G Y = 137122.9406\ N$$

Bolt load under tight pressure

$$W_{m_2} = \underbrace{2\pi b G m P}_{\text{Total joint contact surface compression load } (H_p)} + \underbrace{\frac{\pi}{4} G^2 P}_{\text{Total hydrostatic end force } (H)} = 54646.45\ N$$

Controlling load = max (W_{m_1}, W_{m_2}) = W_{m_1}

Min. bolt cross-sectional area,

$$A_m = \frac{W_{m_1}}{f_a} = \frac{137122.9406}{100.6} = 1363.05\,mm^2$$

Bolt details for nominal diameter G (≤ 508 mm)

Thread nominal diameter = M16

Bolt circle diameter (C_b) = 470 mm

[d_3 - column IS : 4864 - 4870 (1968)]

No. of bolts (n_b) = 16

Root diameter (d_{br}) = 18 mm

Actual Bolt cross sectional area,

$$A_b = (\pi/4)\, d_{br}^2 \times n_b = 4071.50\ mm^2 > A_m$$

Min. Bolt cross-sectional area,

$$N_{min} = \frac{A_b f_b}{2\pi y G} = \frac{4071.50 \times 100.6}{2\times\pi\times 52.46\times 397.67} \times \frac{1}{0.101} = 30.94\,mm$$

↳ in kgf/mm²

(N = 35 mm > 30.94 mm)

Flange Thickness

Flange Bolt Load (W) = $\begin{cases} \left(\frac{A_m + A_b}{2}\right) f_a, & \text{for seating cond.} \\ W_{m_2}, & \text{for operating cond.} \end{cases}$

Gasket Seating cond. (no internal load applied)

$$W = \left(\frac{A_m + A_b}{2}\right) f_a = \left(\frac{4071.50 + 1816.48}{2}\right) \times 100.6$$

$$= 296165.394\ N$$

Flange Moment (Gasket seating cond.)

$$M_F^o = \frac{W(C_b - G)}{2} = \frac{296165.39\,(470 - 397.671)}{2}$$

$$\therefore M_F^o = 10710678.25\ Nmm$$

For operating cond.,

B = Center line to center line bolt spacing
$= D_c = 397.3$ mm

Hydrostatic end force on area inside of flange, $H_p = \frac{\pi B^2 P}{4} = 47109.73$ N

So, $h_D = \frac{C_b - B}{2} = \frac{470 - 397.3}{2} = 36.35$mm

Moment due to H_p,
$M_D = H_D h_D = 1712433.686$ N-mm

Gasket load under operating cond.,

$$H_G = W - H$$

$H = \frac{\pi G^2 P}{4} = 47183.27$ N , $W = W_{m_2} = 54646.45$ N

$\therefore H_G = 7463.18$ N

$$h_G = \frac{C_b - G}{2} = \frac{470 - 397.61}{2} = 36.195\text{mm}$$

Moment due to H_G is,
$M_G = H_G h_G = 270129.80$ N-mm

Pressure force on flange face, $H_T = |H - H_D|$
$= 39646.55$ N

$h_T = \frac{h_D + h_G}{2} = 36.273$ mm

Moment due to H_T,
$M_T = H_T h_T = 1438079.485$ N-mm

Summation of flange moments for operating cond.,

$M_F = M_D + M_G + M_T = 3420642.971$ N-mm

Under seating cond., $M_F^\circ = 10710673.25$ N-mm

Since, $M_F^\circ > M_F \Rightarrow$ Controlling moment $M = M_F^\circ$

Flange Thickness

A = 510 "d_2 column" IS: 4864 - 4870 (1968)

$$K = \frac{C_b + 2E}{B} = \frac{A}{B} = \frac{510}{397.3} = 1.284$$

For flange, $Y = f(K)$ Fig 12.2
[PED Brownwell L.E ; E.H Young]

$$Y = \left| \frac{1}{K-1} \left[0.66845 - 5.71690 \frac{K^2 \log_{10} K}{K^2 - 1} \right] \right|$$

On graph-fitting, $Y = 8$

$$t_F = \sqrt{\frac{M_F^o Y}{f_E B}} = \sqrt{\frac{10710673.25 \times 8}{(100.6)(397.3)}} = 46.3 \text{ mm}$$

$\therefore t_F = 46.3$ mm

Supports

Horizontal Heat Exchanger unit

⇒ 2 Supporting saddles with holes for anchor bolts
⇒ Holes in at least one support shall be elongated to provide for expansions of shell.

Code Output

```
Design parameters of a Shell & Tube Heat Exchanger

Hot Fluid: Heavy Naphtha
Cold Fluid: Water
Enter the mass flow rate of Heavy Naptha in kg/s(if you want to enter in
1.5+0.03*G format enter 'G'): G
Group Number: 29
--------------------------------------------------

Mass Flow Rate of Heavy Naptha: 2.37 kg/s
Please input the following values in degree Celsius.
Initial Temperature of the Hot Fluid: 85
Desired Final Temperature of the Hot Fluid: 55
Coolant initial temperature: 35
Coolant final temperature: 45

--------------------------------------------------

Caloric Fraction of Hot Fluid: 0.4348
Caloric Temperature Of Hot Fluid: 68.0429 C
Caloric Fraction of Cold Fluid: 0.3774
Caloric Temperature Of Cold Fluid: 38.7737 C

--------------------------------------------------

Properties of the Fluids at Tc

Hot Fluid: Heavy Naptha
Specific heat Capacity of Hot Fluid: 2036.9947 J/kg·K
Viscosity of Hot Fluid: 0.7333032907813091 cP
Denisty of Hot Fluid: 791.3870 kg/m³
Thermal Conductivity of Hot Fluid: 0.1500 W/m·K

Cold Fluid: Water
Specific Heat Capacity of Water: 4178.5761 J/kg·K
Viscosity of Cold Fluid: 0.6681 cP
Denisty of Cold Fluid: 992.6812 kg/m³
Thermal Conductivity of Cold Fluid: 0.5980 W/m·K

--------------------------------------------------
```

Enthalpy Heat Balance

Heat Rate: 144830.3237 J/kg·s
Mass Flow Rate of Water: 3.4660 kg/s

LMTD and LMTD Correction Factor

LMTD considering counter-current flow: 28.8539 C
R: 3.0 S: 0.2
LMTD correction Factor for 1 shell pass and 2 or more tube passes
Ft = 0.9350
LMTD correction Factor for 2 shell pass and 4 or more tube passes
Ft = 0.9847
1-2 Heat Exchanger is the better option. As Ft>=0.9. The code will proceed calculating parameters for 1-2 HE.

Shell Side and Tube Side Parameters

<u>Iteration 0</u>
U(assumed): 496.8480 $W/m^2 \cdot K$
Area: 10.8043 m^2
Required number of Tubes: 23
Standard Number of tubes: 32
Ds_sq:0.2540 m

Tube Side Fluid: Heavy Naptha
Tube Side Reynolds Number (Re|tube): 15112.8485
Tube Side Heat Transfer Coefficient (ho):890.9150 $W/m^2 \cdot K$

Shell Side Fluid: Water
Shell Side Reynolds Number (Re|shell): 20210.1467
Shell Side Heat Transfer Coefficient (ho): 1134.0369 $W/m^2 \cdot K$

Ucal: 244.7841 $W/m^2 \cdot K$
Relative error = 50.7326 %
Since the relative error is >5% we proceed to do further iterations

<u>Iteration 1</u>
Required number of Tubes: 46
Standard Number of tubes: 45
Ds_sq:0.3048 m

Tube Side Reynolds Number (Re|tube): 10746.9145

```
Tube Side Heat Transfer Coefficient (hi):678.2444 W/m²·K
Shell Side Reynolds Number (Re|shell): 14034.8241
Shell Side Heat Transfer Coefficient (ho):  1185.5418 W/m²·K
Ucal: 214.0128 W/m²·K
Relative error =  12.5708 %
```

<u>Iteration 2</u>

```
Required number of Tubes: 52
Standard Number of tubes: 56
Ds_sq:0.3365 m

Tube Side Reynolds Number (Re|tube): 8635.9134
Tube Side Heat Transfer Coefficient (hi):569.3847 W/m²·K
Shell Side Reynolds Number (Re|shell): 11511.6535
Shell Side Heat Transfer Coefficient (ho):  1066.4834 W/m²·K
Ucal: 189.8284 W/m²·K
Relative error =  11.3004 %
```

<u>Iteration 3</u>

```
Required number of Tubes: 59
Standard Number of tubes: 76
Ds_sq:0.3873 m

Tube Side Reynolds Number (Re|tube): 6363.3046
Tube Side Heat Transfer Coefficient (hi):445.9697 W/m²·K
Shell Side Reynolds Number (Re|shell): 8690.2002
Shell Side Heat Transfer Coefficient (ho):  918.0437 W/m²·K
Ucal: 159.1453 W/m²·K
Relative error =  16.1636 %
```

<u>Iteration 4</u>

```
Required number of Tubes: 70
Standard Number of tubes: 76
Ds_sq:0.3873 m

Tube Side Reynolds Number (Re|tube): 6363.3046
Tube Side Heat Transfer Coefficient (hi):445.9697 W/m²·K
Shell Side Reynolds Number (Re|shell): 8690.2002
Shell Side Heat Transfer Coefficient (ho):  918.0437 W/m²·K
Ucal: 159.1453 W/m²·K
Relative error =  0.0000 %
```

Pressure Drop and Overdesign

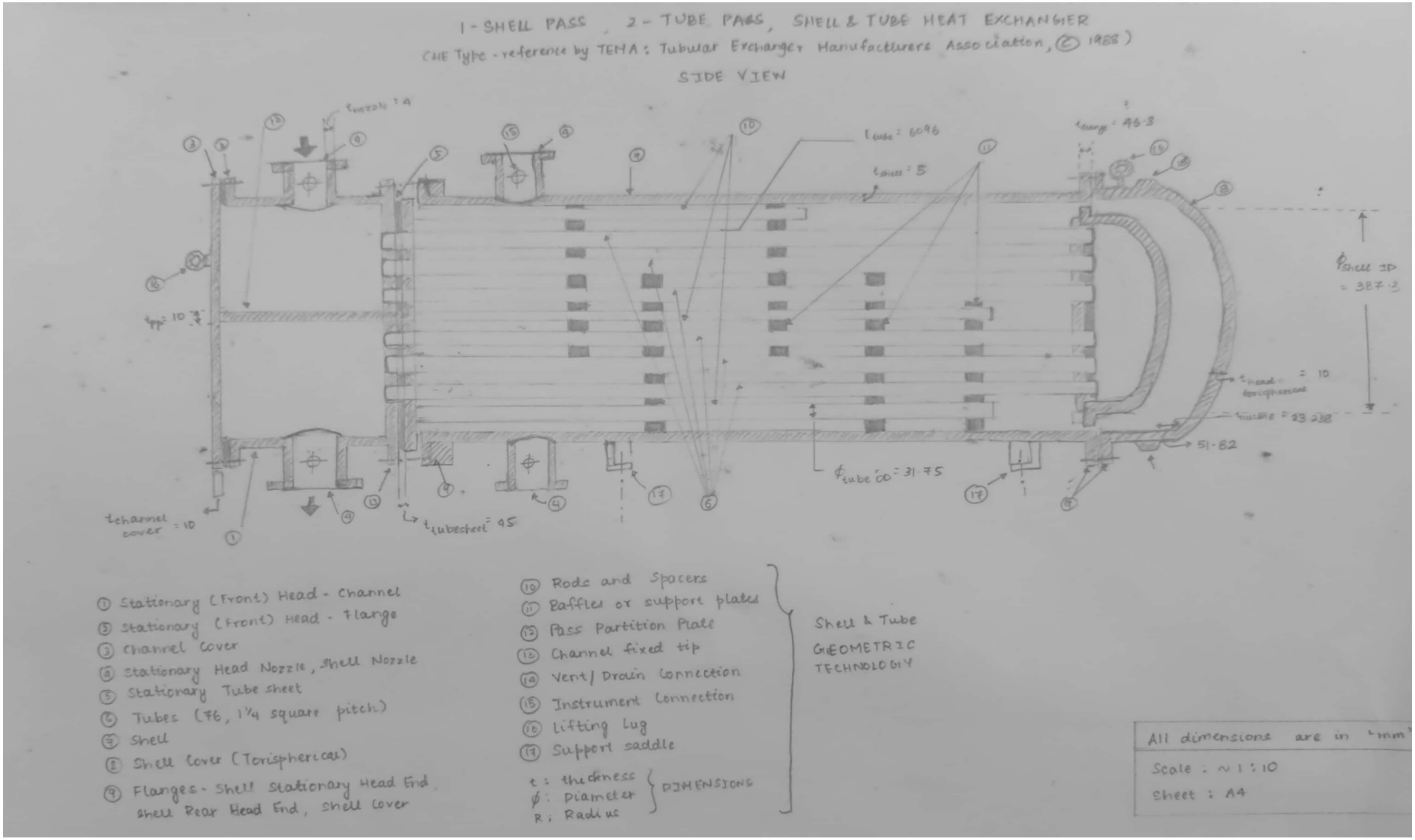
```
Tube side velocity: 0.3465 m/s
```

```
Friction Factor: 0.04462
delta Pf:759.1828 Pa , delta_Pr: 0.061871 Pa
Tube Side Pressure Drop: 0.1101 psi ( 759.2446 Pa )
Required Number of Tubes: 70
Standard Number of Tubes: 76
%Overdesign: 8.5714 %
```

Design is acceptable and cost-effective

References

1. Process Heat Transfer Principles and Applications by Robert. W. Sarat
2. Process Heat Transfer by D.Q.Kern
3. India Standard Specification for Shell and Tube Type Heat Exchanger (IS: 4503-1967)
4. Standards of the Tubular Exchange Manufacturers Association
5. NPTEL: MECHANICAL DESIGN OF HEAT EXCHANGER
   https://nptel.ac.in/content/storage2/courses/103103027/pdf/mod2.pdf
6. Other Links:
   a. Nozzle Thickness
      https://nptel.ac.in/content/storage2/courses/103103027/pdf/mod2.pdf
   b. API of Heavy Naptha
      https://inside.mines.edu/~jjechura/Refining/02_Feedstocks_&_Products.pdf
   c. Properities of Heavy Naptha
      i. https://m.cameochemicals.noaa.gov/chris/PTN.pdf
      ii. http://crudemarketing.chevron.com/cr
   d. Properities of Water
      http://www.vaxasoftware.com/doc_eduen/qui/caloresph2o.pdf

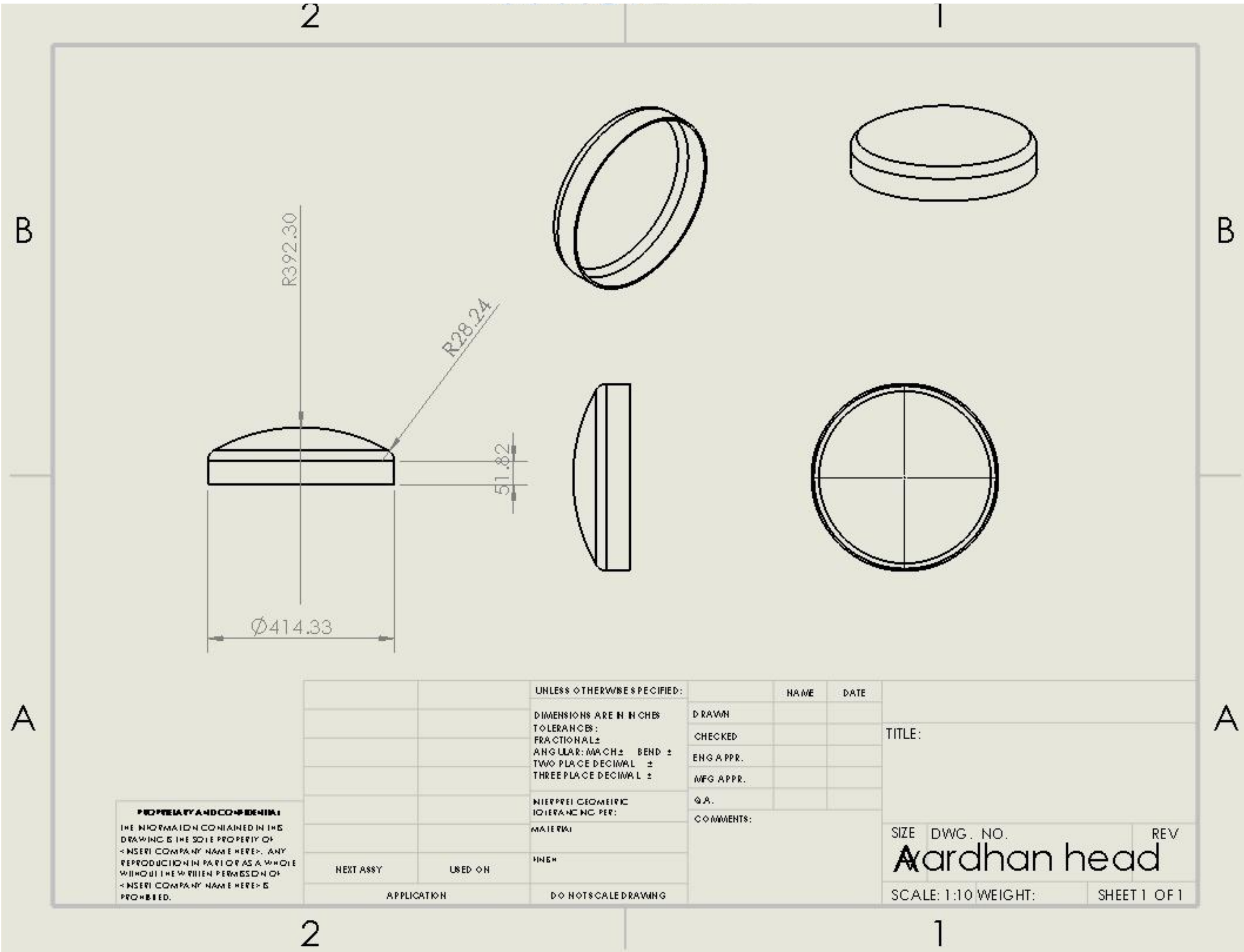


1 - SHELL PASS , 2 - TUBE PASS, SHELL & TUBE HEAT EXCHANGER
CHE Type - reference by TEMA : Tubular Exchanger Manufacturers Association, © 1988)
SIDE VIEW
t_{tube} : 6096
t_{shell} : 5
t_{flange} : 46.3
$\phi_{shell\ ID}$ = 387.3
t_{pp} : 10
$t_{head\ torispherical}$ = 10
51.82
$\phi_{tube\ OD}$ = 31.75
$t_{channel\ cover}$ = 10
$t_{tubesheet}$ = 45
1 Stationary (Front) Head - Channel
2 Stationary (Front) Head - Flange
3 Channel Cover
4 Stationary Head Nozzle, Shell Nozzle
5 Stationary Tube sheet
6 Tubes (76, 1¼ square pitch)
7 Shell
8 Shell Cover (Torispherical)
9 Flanges - Shell Stationary Head End, Shell Rear Head End, Shell Cover
10 Rods and Spacers
11 Baffles or support plates
12 Pass Partition Plate
13 Channel fixed tip
14 Vent/ Drain Connection
15 Instrument Connection
16 Lifting Lug
17 Support saddle
t : thickness
ϕ : Diameter
R : Radius
DIMENSIONS
Shell & Tube GEOMETRIC TECHNOLOGY
All dimensions are in 'mm'
Scale : ~ 1 : 10
Sheet : A4

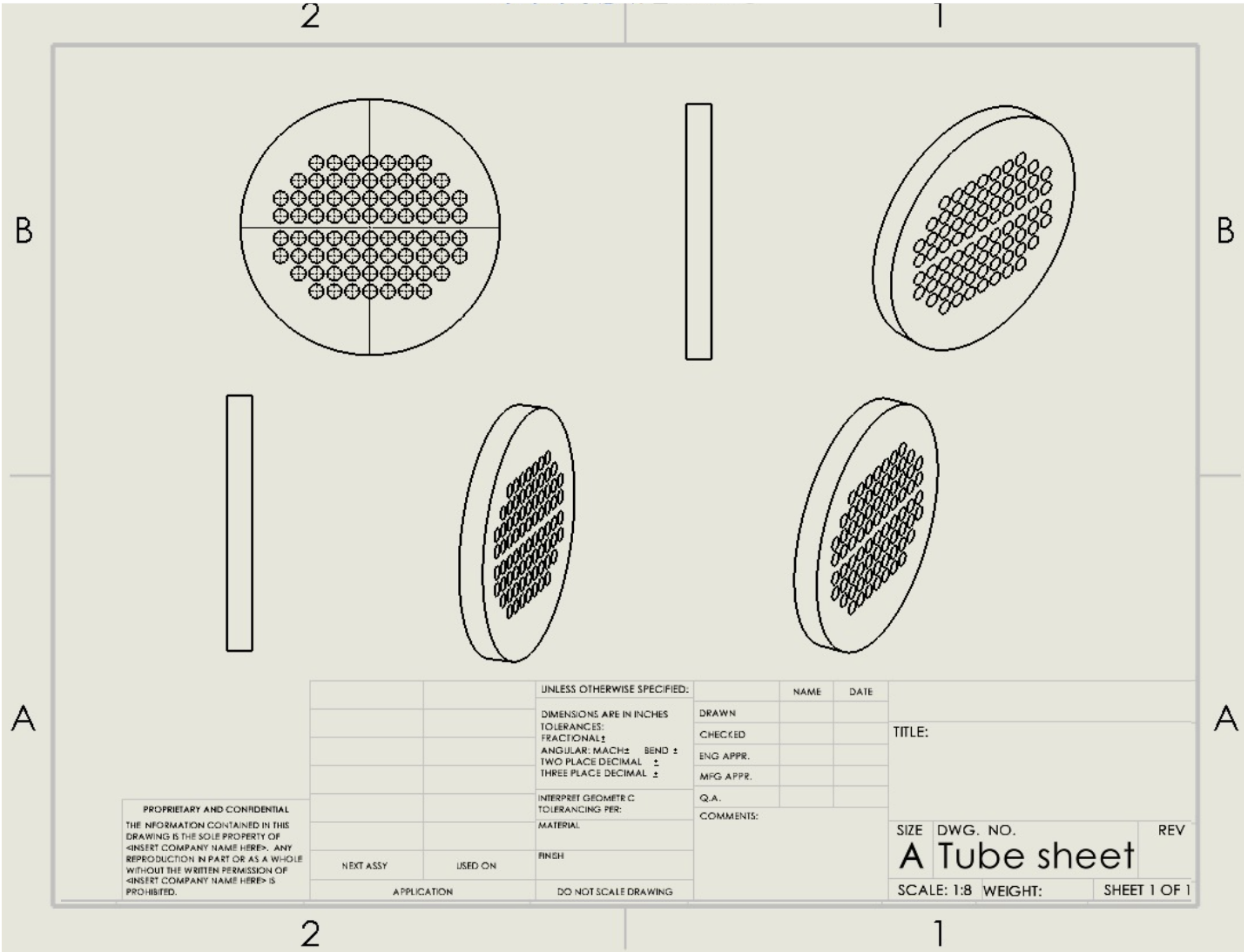
SOLID WORKS DESIGN FILES

TORISPHERICAL HEAD
TUBESHEET LAYOUT
SHELL AND NOZZLE
FLANGE

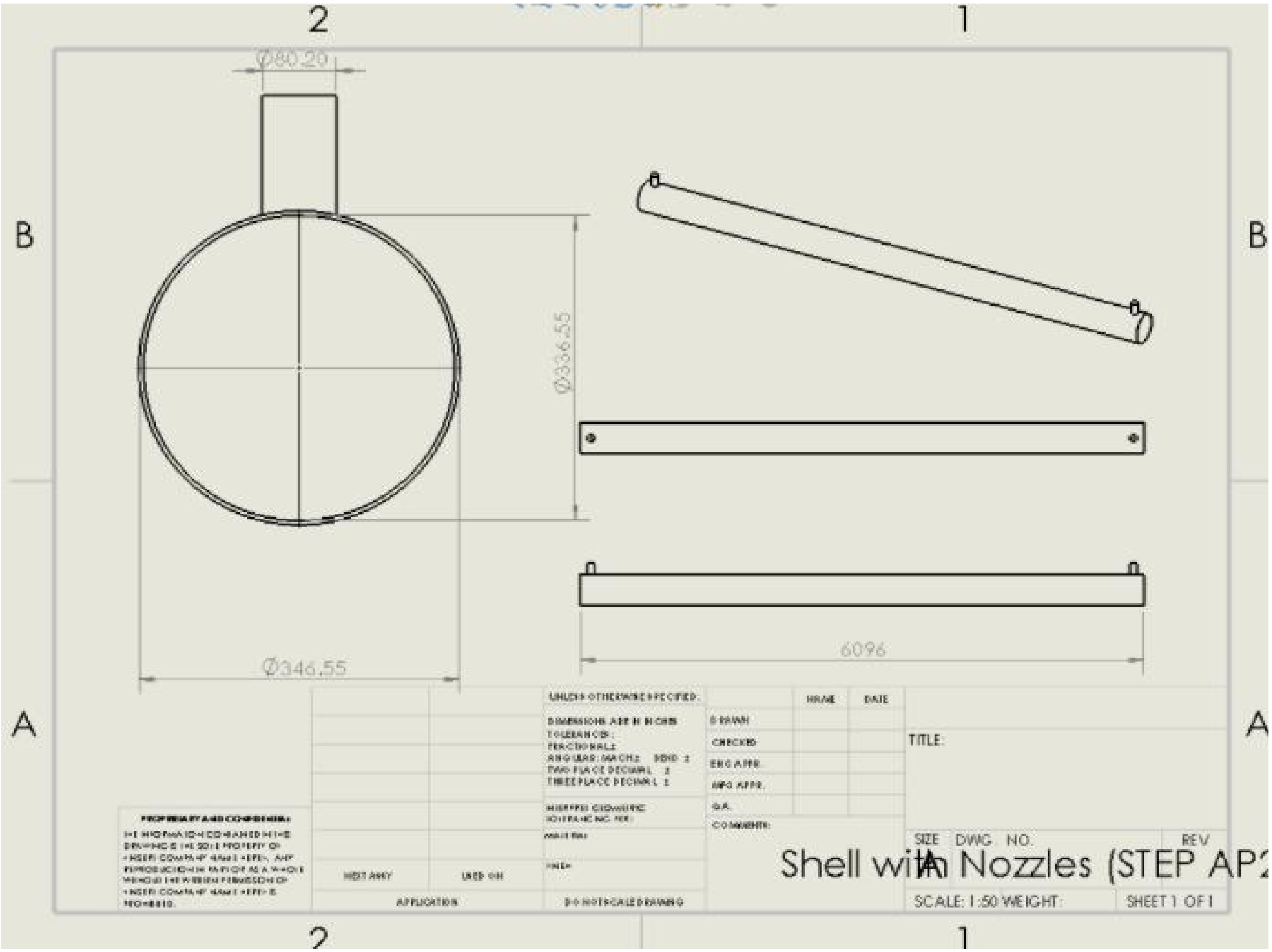
TORISPHERICAL HEAD



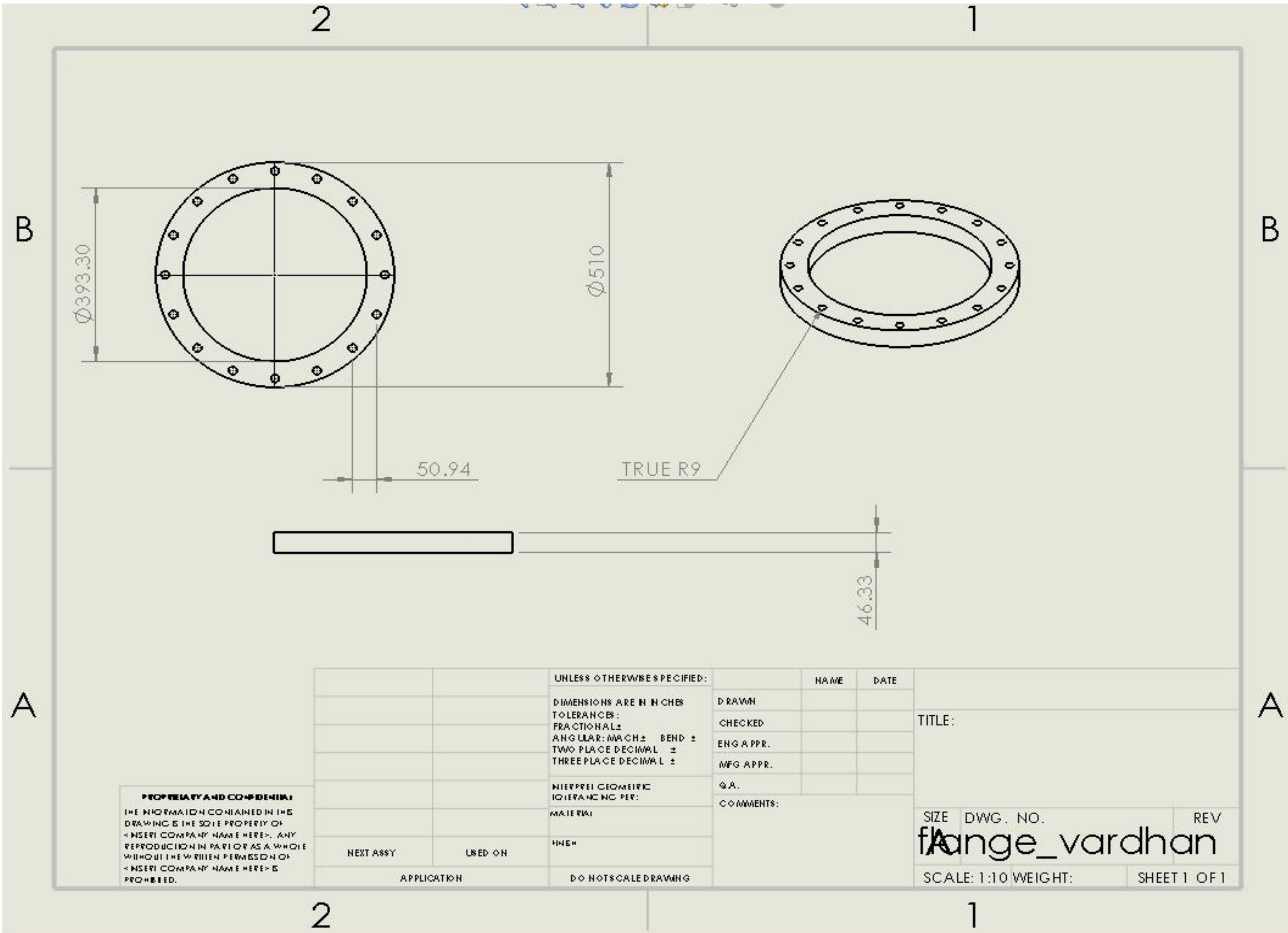
TUBESHEET LAYOUT



SHELL AND NOZZLE



FLANGE


2
1
B
A
Ø393.30
Ø510
50.94
TRUE R9
46.33
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
MATERIAL
FINISH
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
PROPRIETARY AND CONFIDENTIAL
SIZE
DWG. NO.
REV
A
flange_vardhan
SCALE: 1:10 WEIGHT:
SHEET 1 OF 1